Enhanced Condensation for Organic Rankine Cycle

4th Quarterly Progress Report

Reporting Period: Oct 1, 2013 – Dec 31, 2013

Project Grantor: **Alaska Energy Authority**
Alan Baldivieso, Program Manager
813 W. Northern Lights Blvd. AK 99503
907-771-3027 (voice), Email: abaldivieso@aidea.org

Project Grantee: University of Alaska, Institute of the Northern Engineering
Maggie Griscavage
P.O. Box 757880, Fairbanks, AK 99775
907-474-7301 (voice), Email: fygrcon@uaf.edu\

Grant Agreement Number: 7310028
Project Code: 413006

Principal Investigators: **Sunwoo Kim**, PhD (PI)
Mechanical Engineering Department
University of Alaska, Fairbanks
907-474-6096 (voice), Email: swkim@alaska.edu

1. BACK GROUND

Generating electricity from low grade heat sources has attracted attention due to rising fuel price and increasing energy demand. The organic Rankine cycle (ORC) system is the most practical solution among technologies developed for low grade heat recovery. However, the efficiency of a typical small scale ORC is 10% or less. Most energy loss in the ORC is attributed to thermodynamically irreversible heat transfer processes occurring in its heat exchangers: the evaporator and condenser. In particular for waste heat recovery ORCs, economical success is mainly determined by effectiveness of the condenser because, while their heat source is provided at no cost, heat rejection accounts for most of operation cost. Almost half of total cost for operation and maintenance of an ORC system can stem from its condenser. We investigate and demonstrate heterogeneous condensing surfaces that potentially reduce the irreversibility during the condensation of organic fluids.

2. PROGRESS REPORT

We have made progress during the reporting period (Oct 1 – Dec 31, 2013) and progress activities are described below.

Task 1: Model Development (Completed)

Task 2: Design and Construction of Testing Apparatus

Since the last reporting period, a couple of modifications have been made. First of all, a micro-pump that operates at low flow rates was equipped to the drain line to steadily deliver the condensate accumulated in the condensation chamber back to the evaporation chamber. The pump was connected to a speed controller. The pump RPM is to be manually adjusted such that the level of condensate does not exceed the bottom of the condensing surface and the pressure of the condensation chamber is kept unchanged. After leakage was found from the vacuum line (the tube connecting the condensation chamber to the vacuum pump), we replaced the fitting and tube with a stainless steel tube and compression type fittings. We also added one more resistance heater in the evaporation chamber to upgrade the heating capacity.

The test apparatus has been maintained in operable condition. The modifications made during this report period improved reliability and capacity for initial performance tests.

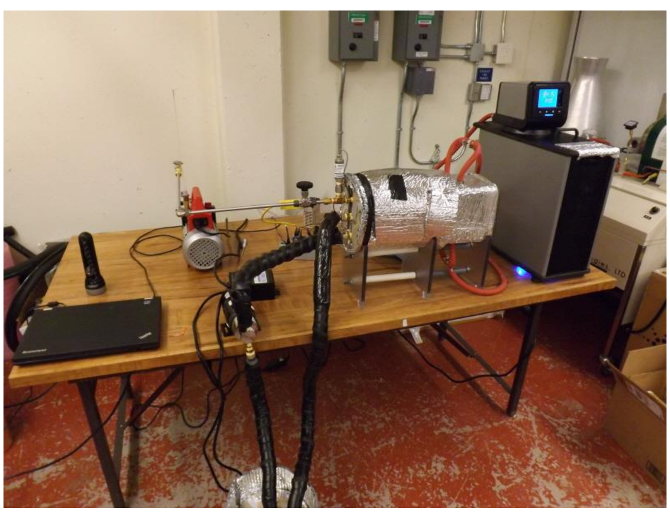

Figure 1: Photograph of test apparatus after a micro-pump equipped in drain line

Figure 2: The micro-pump and RPM controller

Task 3: Initial Performance Testing

Initial performance tests were conducted using deionized water as working fluid at around 100°C and 102 kPa. Data collection procedure and description of apparatus are as follows.

Steam is generated from distilled water in the heating chamber. The steam then flows over the condensing surface (3-inch diameter copper), which is located in the condensing chamber. The condensing surface's temperature is maintained at the required temperature by maintaining the temperature of the coolant, which flows on the other end of the condensing block in the cooling chamber. A chiller pump is used to control the coolant's temperature. The condensate from the test condenser flows down in to the condensing chamber. The condensate is then pumped into the heating chamber to be turned into steam again using a very small pressure differential pump. The vacuum pump is operated from time to time to remove non-condensable gases from the system. This vacuum pump is also used to maintain a set pressure in the condensing chamber. The pump which is used to drain the condensate from the condensing chamber is also used to fine adjust the pressure in the condensing chamber. Thermocouples are inserted in the condensing copper block to find out the surface temperature of the condensing block.

The evaporation chamber has three cartridge heaters, (max 0.9 KW) which are controlled by two variacs. With the help of variacs, we are able to control the total heat output from the heaters. The heating chamber and the condensing chambers are connected with hoses. When the temperature is above 100°C in the evaporation chamber, the steam starts flowing into the condensing chamber through the hose. At this moment the chiller pump is still turned off, which means the condensing block is at room temperature. The steam which flows into the condensing chamber starts condensing on the condensing surface. We let the condensing chamber temperature raise up to 90°C before we turn on the chiller pump. The reason behind this is we want to let the temperature rise in the condensing chamber to be gradual, not exponential. Turning on the chiller pump at lower condensing chamber temperatures would disturb the rise in temperature in the condensing chamber. We try to maintain the condensing chamber at 100°C and at a pressure close to the room pressure. We first adjust the variac so that the temperature is around 100°C and then when the condensing chamber temperature is high enough, we keep adjusting the chiller pump's temperature to get the desired condensing surface temperature.

The condensate level in the condensing chamber is maintained at a certain level by operating the pump. Adjusting the pump's RPM will also affect the pressure in the condensing chamber, because the pump can also move the vapor from condensing chamber to the heating chamber, which decreases the pressure in the condensing chamber. We use a pump to finely control the pressure in the condensing chamber. Presence of non-condensable gases in the system decreases the heat transfer coefficient of the system. Thus, removing of non-condensable gases plays a very important role in determining the right heat transfer coefficient value. We constantly monitor the pressure and the corresponding saturation temperature of the liquid to determine the presence of non-condensable gases. If the saturation pressure doesn't match the saturation temperature, this indicates the presence of non-condensable gases. To tackle non-condensable gases we use a vacuum pump, which is operated manually to remove all the non-condensable gases out from the condensing chamber. We only use the vacuum pump once or twice at the beginning of the experiment. Once all the non-condensable gases are removed from the system, there is no need for the vacuum pump. For the most of the experiment, we keep the vacuum closed.

Controlling temperature and pressure is a very important part of determining the heat transfer coefficient. To obtain the desired ΔT (difference of steam temperature and condensing surface temperature), it is required to control both the variac and the chiller pump temperature. We set the input voltage to the heaters with the help of the variac, such that the condenser chamber's temperature is 100°C. The chamber temperature should always be 100°C. Thus, using trial and error, we figured out the chiller pump's temperature to get the desired condensing surface temperature. In this process the temperature of the condensing chamber keeps changing, consequently we keep adjusting the input heat to the heaters by changing the variac's settings. The temperature of the condensing chamber changes the pressure also changes. We try to maintain the pressure of the condensing chamber at atmospheric pressure with the help of the pump and the variacs.

We wait a period of time after changing each individual setting, giving enough time to the system to respond to the changes. It is very important that the system is at equilibrium before we measure the heat transfer coefficient. Giving the system a long enough time after any changes made to either of the above two mentioned devices and seeing no further changes in temperature and pressure, we start monitoring the pressure and temperatures closely for another half an hour. In this half an hour time we make sure the temperature and pressure keep unchanged. If there is any increase or decrease we adjust the settings in the variacs or pump, so that the pressure oscillates within ± 0.1 psi range from atmospheric pressure. By doing this the temperature is also controlled. Taking the average pressure and temperature reading for the last half an hour would give us one set of data for a specific ΔT. We then change the variac and chiller pump settings to find out temperature and pressure values at different ΔT.

<u>Measuring and acquiring Data</u>

We use hermitically sealed K type thermocouples which were manufactured by Omega. We use sealed tip thermocouples to avoid any contact of the thermocouple tip with the condensing block. Otherwise, a junction of three metals forms at the intersection, which affects the temperatures we measure. We use an Omega pressure transducer which is capable of measuring pressures from 0 to 30 psi. NI DAQ to acquire the data from the thermocouples and the pressure transducers is connected.

<u>Determination of heat transfer coefficient</u>

The condensing block is insulated, not allowing any heat transfer from the condensing block to the environment. That is when the radial heat transfer from the condensing block is negligible. Then we only consider the heat is being travelling in only one direction that is from the condensing chamber to the cooling chamber. One-directional conduction analysis is used to find out the surface temperature of the condensing block, and the heat transfer coefficient. For measuring the heat transfer coefficients and the surface temperature of the condensing block, we find out the temperature distribution in the condensing block and then extrapolate the values to find the condensing surface temperature.

Local heat flux (q_i): $k\frac{\Delta T_i}{L}$

Average heat flux (q): $\frac{1}{3}\sum_{i=1}^{3} q_i$

Surface temperature (T_s): $T_1 + \left(q\frac{L}{k}\right)$

Heat transfer coefficient (h): $\frac{q}{A\Delta T}$

where ΔT_i is the temperature difference of two thermocouple

q_i is the local heat flux

q is the average heat flux

T_s is the surface temperature

T_1 is the temperature reading of closest thermocouple from the surface

L is the distance between thermocouples

A is the surface area of the condensing surface.

Results of initial performance test

The experiments are carried out for ΔT 3-10°C to check whether the test apparatus generates reasonable data (ΔT is the temperature difference between the vapor in condensation chamber and the condensing surface temperature). The results obtained for the heat transfer coefficients at a steam pressure of 15 psi are shown in Figure 3. In this figure the experimental heat transfer coefficient values are plotted in a curve in blue. The theoretical heat transfer coefficients that are calculated based on the Nusselt theory of condensation are also plotted in red. The trend of the experimental data looks similar to that of theoretical values in that the heat transfer coefficient decreases for a large temperature difference. However, the experimental heat transfer coefficients are measured to be greater than that of the theory. The theoretical heat transfer coefficient is assumed that the entire condensing surface is covered by a film of condensate. In order words, mode of theoretical condensation is filmwise condensation. However, in experiments conducted, we observed a locally dropwise condensation. This partial dropwise condensation explains the main reason behind the "*h*" values being much higher than theoretical "*h*" values. It is also observed that more area is occupied by dropwise condensation mode for a low ΔT. This may explain the reason that the discrepancy is large for low temperature differences and gradually decreases.

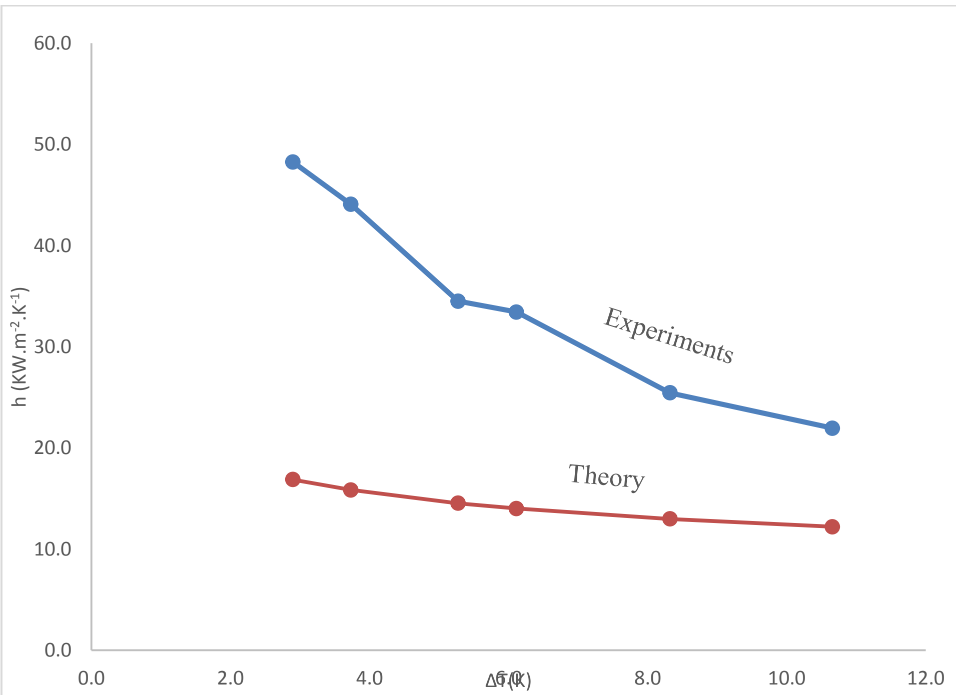

Figure 3: experimental *h* values (condensation heat transfer coefficient) in comparison with theoretically calculated *h* values